Concept modelling of automotive beams, joints and panels

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Abstract: - The paper proposes a methodology for the concept design of beam-like structures, joints and panels in a vehicle FE model, with the aim of enabling accurate NVH simulations of the Body in White (BIW) already in the initial phases of the vehicle design process. Concept models of beams and joints are created, respectively, by means of a geometric analysis of beam-member cross-sections and a static analysis of joints. Concept panels are modelled by coarsening the original FE mesh, while preserving the characteristics of these parts in terms of basic geometric features, such as shape and curvature. The proposed approach is illustrated by using an industrial case study, where simplified models of beams, joints and panels of the upper region of a vehicle’s BIW are created and validated through full vehicle FE analyses. In line with the standards used by automotive manufacturers, two static load cases are defined and analyzed to assess the accuracy of the concept model in terms of static torsion and bending stiffness of the vehicle body.

Key-Words: Vehicle Body, Concept Design, Finite Element Method, Vehicle Stiffness, Automotive Engineering, Model Reduction

1 Introduction

More than ever before, industries that operate in highly competitive environments have to face the complex problem of developing products of increasing quality at affordable costs. Such a challenge is typical in technological fields, where product innovation is to be promoted with the highest priority in order to be successful under the competitive pressure of global markets. The considerations above fully apply to automotive manufacturers, which are forced to develop new or renewing existing products that meet conflicting demands from customers and regulatory bodies, in shorter and shorter timeframes. The objective of improving functional performance attributes of a vehicle, such as safety, noise and vibration, comfort and handling, ecological impact, etc…, while fulfilling a complex set of technological requirements, can only be achieved when engineers are able to analyze what-if cases in any phase of the vehicle development process from the concept stage onwards.

In the last decades, several methodologies have been developed for the concept design of vehicles, with the aim of enabling the use of CAE techniques already in the earliest phases of the design process [1–5], when functional performance targets and industrial requirements are defined, but detailed geometrical data are not yet available. The common objective of all concept design methodologies is to improve the initial CAD design, thus shortening the vehicle development cycle and hence the time-to-market. Several approaches have been proposed. One class of methods uses a predecessor model for the creation of the concept model. If an existing FE model is available and an improved variant is the objective, Mesh Morphing and Concept Modification Approaches can be employed [6-8]. In case a predecessor FE model is not available, the design of a completely new car concept can be supported already in the earliest phases by methods “from scratch”. This second class of methods includes the Topology Design Optimization [9-11] and the Functional Layout Design [12]. By enabling pre-CAD predictions, methods from scratch are useful to identify issues and include possible countermeasures already in the initial CAD design. A third class of concept design methodologies is represented by the methods concurrent with CAD, which provide simulation results as soon as component-level CAD models are available [13]. In the field of methods from predecessor models, Donders et al. [14] proposed a concept modelling...
methodology based on predecessor FE models, namely the “Reduced Beam and Joint Modelling”, with the aim of enabling a fast NVH optimization of an existing vehicle body. This approach, which is implemented in the commercial software program LMS Virtual.Lab [15], allows the creation of concept beams and concept joints, consisting of two-node FE beams and joint superelements respectively. A modal reduction of the vehicle is then made, and subsequently one can add concept beam and joint modifications to the small-sized reduced modal model, allowing efficient modification of the body beam-like structures and joint connections, sensitivity analysis and efficient optimization of the full-vehicle model.

Recently, Mundo et al. [16] created a reduced model where a number of beam members and joints of the predecessor FE model are replaced by concept models. In this paper, the above mentioned beam and joint concept modelling approach is extended into a concept modelling technique that enables the designer to include two-dimensional structural elements, i.e. panels, in the vehicle concept model. The outline of the paper is as follows. In section 2, the beam and joint concept modelling approach is summarized [16]. The same section illustrates the proposed methodology for concept modelling of panels. An industrial case study is illustrated in section 3. Beams, joints and panels of the upper-front region of a vehicle BIW are replaced by concept models. A static comparison between the original and the simplified model is performed by evaluating the static stiffness of the full vehicle model under torsion and bending, with the aim of providing a validation of the proposed concept modelling approach.

2 Concept modelling of beams, joints and panels

The proposed approach for vehicle body concept design can be summarized as follows: after a region of the vehicle is selected as target of the concept design, panels are removed and a beam and joint layout is identified. A reduced model of the vehicle, where the original mesh of beams and joints is replaced by concept models, is created and validated [16]. Starting from the geometry of the new model, concept panels are created by coarsening the original FE mesh, while preserving the original shape and curvature. This reduced concept model can be used for an NVH optimization of a vehicle body in the initial phases of the design process, before a detailed CAD model of the structure is created.

This section provides an overview of the procedure that is used to estimate the mass and stiffness properties of the simplified beam and joint models [14, 16]. A methodology that enables the inclusion of simplified panel models in the concept design is illustrated as well.

2.1 Concept beams

In a vehicle body, beam-like members can be defined as structures with one dimension (i.e. the longitudinal direction of the beam) that is much larger than the characteristic dimension of the cross-sectional area.

In the FE model of a vehicle, beam-like members are typically thin-walled structures, formed by two or more curved panels, which are usually connected to each other by means of spot welds. In order to replace the detailed mesh of such components by simplified beam elements, a number of beam cross-sections are considered and the equivalent beam properties are computed for each of them. For this purpose, the following procedure is used:

- for each pair of consecutive cross sections, an axis system is created to define the approximate beam direction and a transversal plane;
- by analyzing the shell elements located at each end of the primary member, the two beam centre nodes are identified as the geometric centres of the original cross section;
- these nodes are connected by a linear two-node beam element, i.e. a Nastran CBEAM element, whose stiffness properties (cross-section area and moments of inertia) are calculated by computing and summing the contributions of all the shell elements that belong to each cross-section;
- after such a geometric calculation is performed, the stiffness parameters of each beam-member need a correction that takes into account section variations and discontinuities (holes, spot welds, stiffeners) [17, 18]. For this purpose, correction factors are introduced, whose values are estimated by means of an iterative model updating procedure, which is described in detail in [16].

In the vehicle concept design, the original mesh of the beam-members is removed and replaced by a series of equivalent concept beams. The first and the last beam of each series are connected to the rest of the structure by interpolation elements (Nastran RBE3 superelements), which relate the displacement components of the beam center node to all nodes of the shell elements that are located at the cross section of the original model. The
structural continuity of the concept model is thus guaranteed, as illustrated in Fig. 1-a), which shows the concept models of three beam members along with the original mesh of the joint that connects them.

2.2 Concept joints
In the vehicle concept modelling methodology proposed here, simplified beams are connected to each other by concept joint models. Once a joint is identified as that part of the body where concept beams are connected, a joint group is created that consists of the original mesh, delimited by the end sections of all converging beams, and includes the terminal nodes of each concept beam as well as the interpolation elements. The group is isolated from the rest of the structure and the Guyan reduction is used to calculate a small-sized representation of the joint [19].

Guyan Reduction [19], also known as static condensation, is a method to reduce the FE model of structures into a reduced description in terms of the stiffness and mass matrices condensed at the end nodes. For an arbitrary structure, the basic static FE matrix equation is given by

\[ K \cdot x = F \]  

where \( K \) is the stiffness matrix, \( F \) and \( x \) are the force and the displacement vectors, respectively. By identifying \( n_t \) boundary degrees of freedom (DOFs), which must be retained in the solution, and \( n_o \) internal DOFs, which are to be removed by static condensation, the system of eq. (1) can be partitioned as follows:

\[
\begin{bmatrix}
K_{oo} & K_{ot} \\
K_{to} & K_{tt}
\end{bmatrix}
\begin{bmatrix}
x_o \\
x_t
\end{bmatrix}
= \begin{bmatrix}
F_o \\
F_t
\end{bmatrix}
\]  

(2)

where subscripts \( t \) and \( o \) are used to designate the boundary and the internal DOFs, respectively. From the first line of eq. (2), the internal displacement vector can be determined as

\[ x_o = K_{oo}^{-1} \left( F_o - K_{ot} \cdot x_t \right) \]  

(3)

By introducing the static reduction matrix \( G_{ot} = -K_{oo}^{-1}K_{ot} \) and substituting eq. (3) into the second line of eq. (2), the following equation is obtained:

\[ K_{tt,red} \cdot x_t = F_{t,red} \]  

(4)

where \( F_{t,red} = F_t + G_{ot}^T F_o \) is the reduced loading vector, while \( K_{tt,red} = K_{to}G_{ot} + K_{tt} \) is the \( n_t \times n_t \) reduced stiffness matrix. Physically this matrix represents the stiffness values between each pair of boundary DOFs. This way, the stiffness of the structure has been condensed to the boundary DOFs.

The same transformation can be used to condense the mass matrix on the boundary DOFs, so that an efficient reduced representation of the structure, useful for dynamic analyses, is obtained. However, while exact for the stiffness matrix, the Guyan reduction is an approximation for the mass matrix. By reducing the mass matrix, it is assumed for the considered structure that inertia forces on internal DOFs are less important than elastic forces transmitted by the boundary DOFs. This is true for very stiff components or in cases where local dynamic effects can be ignored. Therefore, the accuracy of the result is case dependent. For typical automotive joints, the stiffness relations between the joint end-sections have a much stronger influence on the global behaviour of the body than the exact distribution of mass along the joint. For this reason, Guyan reduction of the joint structure to its joint end-nodes (i.e. beam center nodes) is an appropriate choice to create a small-sized representation of the actual joint [14].

Each isolated joint model undergoes a static condensation, where the terminal beam centre nodes represent the boundary DOFs that are retained in the solution. As shown in Fig. 1 b), a
concept FE model of the joint is thus created as a small superelement (DMIG), consisting of a reduced stiffness and mass matrix.

2.3 Concept panels
The global modes of a vehicle structure are dominated by the primary beam and joint structure. However, the panels are an important contribution to the body stiffness as well. Therefore, also two-dimensional panels should be included in the concept model of the car. For this reason, a procedure for the creation of panel concept models is proposed. As shown in Fig. 2, the starting point of such a procedure is represented by the concept beam layout, which defines the boundaries of the concept panel. By keeping concept beams as edges, a “flat panel” model is created, characterized by a mesh much coarser than in the original model. In order to make the concept model as accurate as possible, basic geometric features of the original model are to be preserved. For this reason, the flat panel undergoes a morphing process [20], which is based on the superposition of the original mesh over the coarse one: by making each node of the new model coincident with the closest node of the original mesh, the flat panel is properly stretched and curved.

A final concept model of the panel, with shape and curvature closely representing those of the original model, is thus created. A point-to-point connection between the concept panel and the surrounding concept beams and joints is finally achieved by rigidly connecting each boundary node of the panel to the closest beam centre node.

3 Industrial case study
Fig. 3 shows the FE model of a vehicle body, formed by over 100 panels, modelled with linear shell elements and assembled by means of about 3000 spot weld connections, that are represented in the FE model by means of Hexa solid elements [15]. In order to illustrate the concept modelling methodology described in section 2, the front-upper region of the vehicle is selected as target of the concept design. This region has a significant influence on the full-vehicle static behaviour. A group of 10 beam-like structures, namely the A and B-pillars and the longitudinal and transversal roof rails, partly labelled in figure 3 as \( B_1 \ldots B_6 \), are replaced by equivalent simple beams.

Four joints, the two labelled as \( J_1 \) and \( J_2 \) along with the two symmetrically arranged with respect to the longitudinal plane of the vehicle, are replaced by static super-elements (i.e., the equivalent mass and stiffness matrices of each joint). Finally, the vehicle concept model is completed by 3 panels, namely the front windshield (\( P_1 \)), the front (\( P_2 \)) and the rear (\( P_3 \)) panels of the vehicle roof, as shown in Fig. 4. These panels are modelled with a coarse mesh, connected to the surrounding concept beams by means of rigid connections.

With the aim of assessing the accuracy of the concept model, in line with the standards used by
automotive manufacturers, two indicators of the full-vehicle static behaviour are defined: the bending and torsion static stiffness. These indicators are evaluated for both the original and the concept body model by using the two load cases shown in Fig. 5. In both cases, the vehicle body is clamped at the rear suspension locations, while two vertical forces are applied at the front suspensions (points A and B in the same figure). Based on the estimation of the vertical displacements $v_A$ and $v_B$ at the excitation points, the bending and torsion stiffness $K_b$ and $K_t$ of the vehicle body are determined as:

$$K_b = \frac{2FL}{\alpha_b}$$  \hspace{1cm} (5)

$$K_t = \frac{FW}{\alpha_t}$$  \hspace{1cm} (6)

where $F$ is the amplitude of the vertical forces, $L$ is the wheelbase and $W$ denotes the width of the car, while

$$\alpha_b = \arctan \left( \frac{v_A + v_B}{2L} \right)$$  \hspace{1cm} (7)

$$\alpha_t = \arctan \left( \frac{v_A - v_B}{W} \right)$$  \hspace{1cm} (8)

are the deflection angles in the bending and torsion load case respectively.

By means of static FE analyses (Nastran – Sol 101), performed on both the original and the concept model, the stiffness properties of the vehicle body are estimated in accordance with eqs. (5) and (6) and summarized in Table 1. In the same table, the approximation involved by the simplified model with respect to the original model is indicated as well. The results show that both the bending and the torsion stiffness of the original vehicle model is accurately predicted by the concept model, being 0.06% and 0.26% the stiffness overestimation involved by the concept model in the two cases.

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<tr>
<th></th>
<th>Bending</th>
<th>Torsion</th>
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<tbody>
<tr>
<td></td>
<td>Original model</td>
<td>Concept model</td>
</tr>
<tr>
<td>Stiffness (Nm/rad)</td>
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<td>5.04E4</td>
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<tr>
<td>$\Delta %$</td>
<td>--</td>
<td>+0.06</td>
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Table 1: Bending and torsion stiffness of original and concept FE vehicle models

4 Conclusion

A methodology for the concept design of beam-like structures, joints and panels in a vehicle body FE model has been presented in this paper. An industrial case study has been analyzed to demonstrate the proposed approach. The original mesh of ten beams, four joints and three panels from the front-upper part of the vehicle have been replaced by equivalent beam elements, joint super-elements and coarse panels respectively. Two static load cases were defined and analyzed to compare the original and the concept models in terms of full vehicle static stiffness. In the two cases, a good correlation between the simplified and the original models has been obtained, which proves the feasibility of a stand-alone beam-joint-panel replacement layout for static FE analyses.

The final goal of this research is to enable a fast concept optimization of the vehicle body. The stiffness properties of concept beams can be easily modified, which is not directly possible on the complex-shaped cross sections of the original shell mesh.

Before such a goal becomes achievable, however, further validations of the proposed concept design methodology are to be provided by analyzing and
comparing the two models also in terms of dynamic behaviour, which represents one the next steps of the work presented here.

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